Performance Evaluations of a Residential Air Conditioner using R-1132(E) mixed Refrigerants

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ABSTRACT

In the HVAC industry, stringent regulations on refrigerants aim to mitigate global warming, necessitating the replacement of high-GWP hydrofluorocarbon (HFC) refrigerants with lower-GWP alternatives. This study evaluates the drop-in performance of two low-GWP refrigerants: R-474A (GWP < 10) and R-479A (GWP < 150), both containing R-1132(E), in residential air conditioners. We also propose performance enhancement measures for these refrigerants in air conditioning systems and present their estimated effects.

Keywords: R-1132(E), R-474A, R-479A, R-474B, R-479B, Zeotropic Refrigerants

INTRODUCTION

From the perspective of preventing global warming, the greenhouse effect of HFC refrigerants has become a matter of concern, and regulations on the global warming potential (GWP) of refrigerants are being increasingly strengthened. Fig.1 shows the HFC phase-down schedule under the Kigali Amendment to the Montreal Protocol[1], which is a global agreement on the stepwise reduction of HFC refrigerants. For example, developed countries are required to reduce the amount of HFC refrigerants, in terms of CO_2-equivalent, to 15% of the baseline level by 2036. To achieve this level of HFC reduction, it is necessary not only to reduce the refrigerant charge but also to lower the GWP of the refrigerants themselves, thereby necessitating the transition to low-GWP refrigerants.

On the other hand, for the further dissemination of heat pump air conditioners, refrigerants must achieve a balance among safety, environmental performance, energy efficiency, and economic feasibility. However, in the field of air conditioning, no next-generation refrigerant has yet been established that can satisfy all of these requirements.

Goto et al.[2] identified R-1132(E) as a new refrigerant component and developed two R-1132(E)-based refrigerant blends: R-474A, an A2L refrigerant with a GWP below 10, and R-479A, an A2L refrigerant with a GWP of around 150.

This paper reports the results of drop-in performance evaluation tests of a residential wall-mounted air conditioner using R-474A and R-479A, as well as the estimated effectiveness of performance improvement measures required when applying these low-GWP refrigerants to air conditioning equipment.

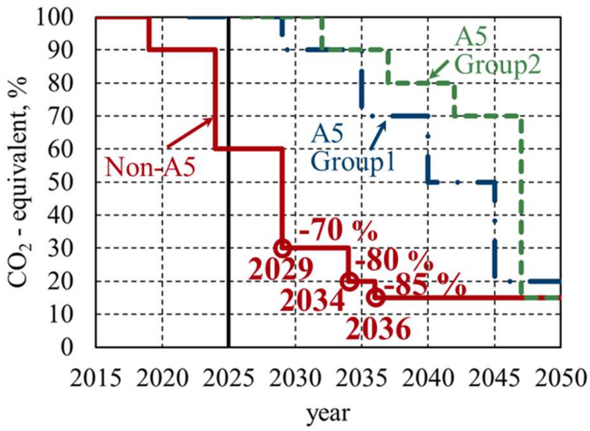

Fig.1 Kigali Amendment

FUNDAMENTAL CHARACTERISTICS OF R-1132(E) MIXED REFRIGERANTS

The calculation conditions for the fundamental characteristics of R-474A and R-479A are presented in Table 1, and the calculated characteristics are shown in Table 2. The GWP values were calculated based on the IPCC Fourth Assessment Report[3], the TEAP Progress Report[4], and the measured values provided by the National Institute of Advanced Industrial Science and Technology (AIST)[5]. Other thermophysical properties were calculated using REFPROP ver.10.0[6], incorporating the mixing rules for R-1132(E) with each refrigerant. In addition, R-474B, which has a different composition ratio from R-474A, and R-479B, which has a different composition ratio from R-479A, are also included.

Table 1 Calculation Conditions

Midpoint Condensing Temperature	[°C]	45
Midpoint Evaporating Temperature	[°C]	14
Condenser Outlet Temperature	[°C]	38
Suction Superheat	[K]	3

Based on Table 2, the potential issues associated with the drop-in application of the above refrigerants to air conditioners designed for R-32 are as follows:

- Degradation of heat exchanger performance due to temperature glide
- Decrease in energy efficiency and cooling/heating capacity resulting from increased refrigerant pressure drop
- Capacity shortage caused by the increased required stroke volume of the compressor

Furthermore, these issues are expected to become more pronounced for refrigerants with lower GWP, as indicated by the characteristics shown in Table 2.

In this study, the drop-in performance of R-474A and R-479A, for which samples were obtained, is evaluated using actual equipment, and the issues are examined.

DROP-IN PERFORMANCE EVALUATION TESTS

These tests aim to assess changes in energy efficiency when replacing the widely used refrigerant R-32 with R-474A or R-479A in a drop-in manner.

For a fair comparison of energy efficiency, the capacity must be matched. Accordingly, based on the refrigerant characteristics in Table 2, the stroke volume of a compressor was adjusted, and it was examined whether equivalent capacity to R-32 could be achieved within the adjustable range of compressor speed.

TEST OVERVIEW

Test Conditions

The Annual Performance Factor (APF), defined in JIS C 9612:2013, was adopted as the energy efficiency index. The test conditions are summarized in Table 3. Although APF calculation requires a low-temperature heating capacity test in addition to these conditions listed in Table 3, this test was omitted because optimization of defrost control is necessary. Instead, the results from the current system's low-temperature heating capacity test were used to calculate the APF.

At each condition, the capacity and the coefficient of performance (COP) were measured.

Table 2 Properties and Performance of Refrigerants

Refrigerant			R-32	R-474A	R-479A	R-474B	R-479B
GWP (AR4)			675	3	147	3	298
Ratio	R-32	[wt%]	100	-	21.5	-	44.0
	R-1132(E)	[wt%]	-	23.0	28.0	31.5	23.0
	R-1234yf	[wt%]	-	77.0	50.5	68.5	33.0
Temperature Glide	Condenser	[K]	0	5.8	4.5	5.9	2.2
	Evaporator	[K]	0	5.0	4.6	5.4	2.2
Theoretical COP Ratio		[%]	100	98.8	97.3	98.2	98.4
Pressure Drop Ratio *		[%]	100	297	191	265	147
Stroke Volume Ratio *		[%]	100	185	134	169	116

(* At the same Capacity)

Table 3 Test Conditions

		Cooling		Heating	
		Rated	Intermediate	Rated	Intermediate
Capacity	[kW]	4.0	2.0	5.0	2.5
Indoor Intake Air Temperature	D.B. [°C]	27		20	
	W.B. [°C]	19		14	
Outdoor Intake Air Temperature	D.B. [°C]	35		7	
	W.B. [°C]	24		6	

Test Units

A residential air conditioner with a rated cooling capacity of 4.0 kW (refrigerant: R-32) was used for testing. The refrigerant circuit (Fig.2) comprised a compressor, four-way valve, outdoor heat exchanger, electronic expansion valve, indoor heat exchanger, and refrigerant line set between the indoor and outdoor units.

Drop-in performance was evaluated for R-32, R-474A, and R-479A, and basically did not change the components throughout the tests. However, to match the capacity for COP comparison, we only changed the compressor when applying R-474A and R-479A. For the compressor modification, we kept the motor specifications the same and only increased the stroke volume by 57%. Refrigerating oil was also changed, and other components were retained after internal cleaning during compressor replacement.

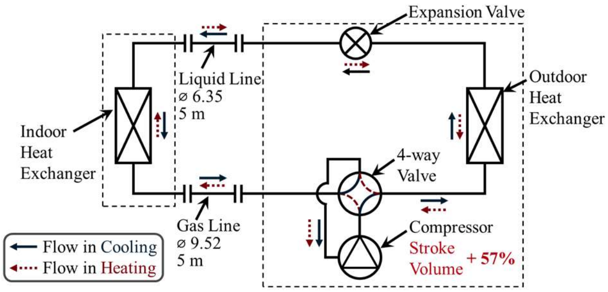

Fig.2 Refrigerant Circuit of the Test Unit

Test Method

The test with R-32 was conducted according to JIS C 9612:2013. For the drop-in tests with R-474A and R-479A, the compressor speed and electronic expansion valve (EEV) opening were adjusted to match the R-32 results for the parameters in Table 4, while other settings, such as blower speed, were kept unchanged.

Table 4 Targets of Actuators

Actuator	Target
Compressor Speed	Capacity
EEV Opening	Superheat at the Evaporator Outlet

TEST RESULTS

Fig.3 presents the drop-in performance evaluation results for R-474A and R-479A. The figure shows the COP values measured under cooling rated, cooling intermediate, heating rated, and heating intermediate conditions, together with the APF calculated from the COP, expressed as ratios relative to R-32.

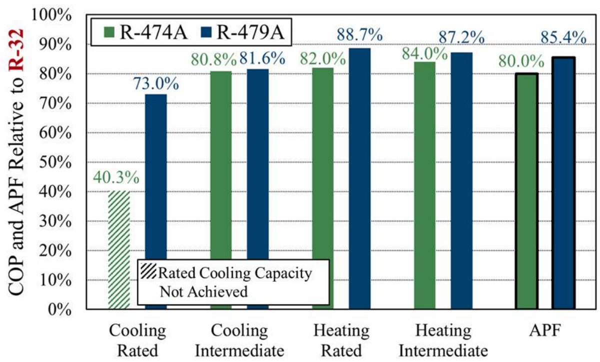

Fig.3 Performance Comparison in Drop-in Tests

For R-474A under the cooling rated condition, equivalent capacity to R-32 was not achieved; details are discussed below. Under the other conditions, equivalent capacity (within ±2% of R-32) was obtained. No frosting of the outdoor heat exchanger was observed under the heating test conditions.

Capacity Shortage in R-474A Cooling Rated Conditions

Under cooling rated conditions, the capacity with R-479A was within ±0.5% of that with R-32, indicating equivalent performance. In contrast, with R-474A, even after actuator adjustments, capacity reached only 89.8% of that with R-32. Accordingly, the APF calculation for R-474A is based on measurement results obtained under rated conditions where the required capacity was not met.

To further investigate this capacity shortage, tests were conducted with the compressor speed varied while the EEV opening was fixed, as shown in Fig.4 and Fig.5. Fig.4 presents the shift in refrigeration cycle operating points on a p-h diagram, while Fig.5 shows changes in cooling capacity and refrigerant density at the compressor suction.

As illustrated in Fig.5, once the compressor speed was high, further increases did not raise capacity, and within the controllable speed range, the same capacity as with R-32 was not achieved. This is attributed to greater refrigerant pressure losses at higher compressor speeds (Fig.4), which lowered suction pressure and, consequently, refrigerant density at the compressor inlet (Fig.5), resulting in insufficient refrigerant circulation.

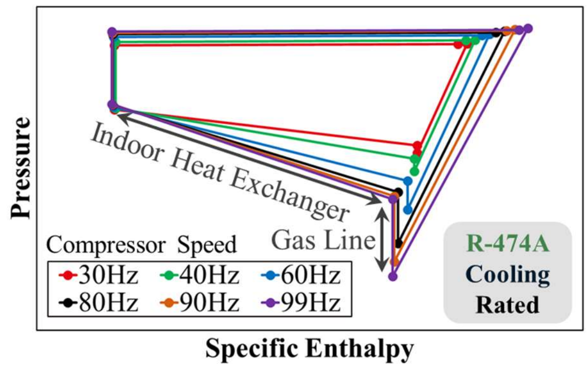

Fig.4 p-h Diagram of Refrigeration Cycles under Various Compressor Speeds (R-474A Cooling Rated Condition)

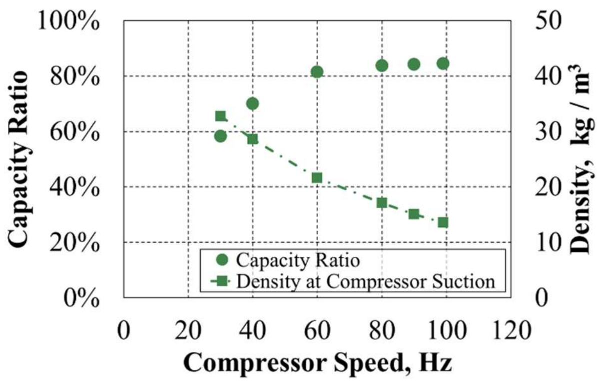

Fig.5 Cooling Capacity and Suction Gas Density vs. Compressor Speed (R-474A Cooling Rated)

ANALYSIS OF PERFORMANCE FACTORS

In drop-in performance tests of R-474A and R-479A, the COP was lower than that of the baseline refrigerant R-32 under all test conditions. To clarify the causes of this degradation, the following factors were identified, and their respective contributions were evaluated:

- Theoretical COP difference due to refrigerant properties: a-1, a-2
- Change in condenser performance: b
- Pressure loss between compressor discharge and condenser: c
- Change in evaporator performance: d
- Pressure loss between evaporator and compressor suction: e
- Adiabatic efficiency of compressor (compressor efficiency): f
- Other factors (e.g., heat dissipation outside heat exchangers, capacity differences): g

Definition of Theoretical COP

In this study, the theoretical COP is defined as the ideal vapor compression cycle without refrigerant pressure drop, heat leaks, compressor losses, and energy consumption other than refrigerant compression. The cycle assumes an isentropic compression process, isobaric heat exchange, and isenthalpic expansion.

Accordingly, the theoretical COP, $COP_{\text{theoretical}}$, is calculated from the refrigerant enthalpy change in the indoor heat exchanger (Δh_{ihex}) and in the compression process (Δh_{comp}) as:

$$COP_{\text{theoretical}} = \Delta h_{\text{ihex}} / \Delta h_{\text{comp}} \quad (1)$$

For the operating conditions used to determine the enthalpy changes in Eq.(1), the condenser outlet temperature and compressor suction temperature were taken from measurements. The condensation and evaporation temperatures were defined as the saturation temperatures corresponding to the mean pressures at the inlet and outlet of the heat exchangers, based on the reference case with R-32.

Condensing and Evaporating Pressures in Theoretical COP

Because R-474A and R-479A are non-azeotropic refrigerant mixtures with temperature glide, their condensing and evaporating pressures cannot be uniquely determined from the corresponding temperatures. In Table 2, refrigerant properties were compared using midpoint temperatures as in Table 1. However, even when midpoint temperatures are the same, the temperature difference between refrigerant and air can vary with flow direction. Consequently, the performance of a heat exchanger matched by midpoint temperature may differ depending on flow configuration and the magnitude of temperature glide.

For the theoretical COP calculation, refrigerant pressures were defined as follows: in the indoor heat exchanger, the pressure was set such that the mean temperature difference between air and two-phase refrigerant matched that of R-32; in the outdoor heat exchanger, the pressure was set such that this difference corresponded to the heat exchange amount required by the refrigerant characteristic. These values were taken as condensing or evaporating pressures. Two flow configurations were considered: (a-1) counterflow in both condenser and evaporator, which increases theoretical COP through the Lorenz cycle; and (a-2) counterflow in the condenser and parallel flow in the evaporator, reflecting flow reversal between cooling and heating. Fig.6 schematically illustrates the procedure used to derive R-474A condensing and evaporating pressures from R-32 experimental data, accounting for refrigerant–air temperature variations and differences.

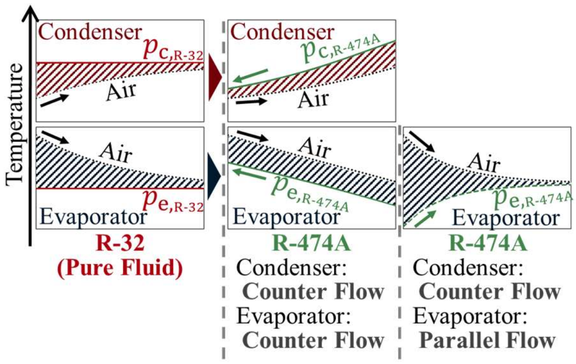

Fig.6 Temperature Changes and Differences

Method of Contribution Analysis for Heat Exchanger Performance, Pressure Drop, and Compressor Efficiency

Fig.7 shows the deviation between an ideal vapor compression refrigeration cycle and the measured operating points, schematically illustrating the effects of heat exchanger performance, refrigerant pressure drop, and compressor efficiency on cycle operation.

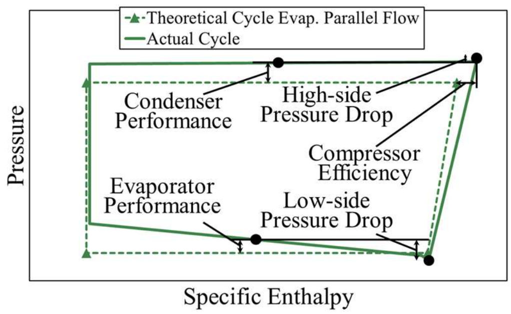

Fig.7 Performance Factors on Refrigeration Cycle

The contributions of these factors to the performance degradation of R-474A and R-479A were evaluated as follows. Here, R_{COP} is defined as the ratio of the COP of the tested refrigerant ($COP_{\text{Drop-in}}$) to that of R-32 ($COP_{\text{R-32}}$), as expressed in Eq.(2):

$$R_{\text{COP}} = COP_{\text{Drop-in}} / COP_{\text{R-32}} \quad (2)$$

1. Calculate the theoretical COP of R-32 and that of the target refrigerant (R-474A or R-479A) assuming counterflow operation of both the condenser and evaporator (a-1), and determine R_{COP}.
2. Recalculate the theoretical COP assuming parallel flow evaporator operation (a-2), and attribute the change in R_{COP} to the evaporator flow configuration.
3. Incorporate each performance variation factor, compute the resulting COP, and attribute the change in R_{COP} to that factor.

<u>Results of Contribution Analysis of Performance Factors</u>

The contribution analysis results for each condition (Figs. 8–11) are summarized as follows:

- With counterflow heat exchangers (condenser and evaporator), the theoretical COP is higher for R-474A and R-479A than for R-32 due to the Lorenz cycle effect.
- When the evaporator is configured in parallel flow, closer to actual equipment specifications, the theoretical COP of both R-474A and R-479A falls below that of R-32.
- Under rated conditions, the high refrigerant circulation rate causes significant performance degradation due to pressure losses on the low-pressure side.
- Under intermediate conditions, the small temperature difference between air and refrigerant leads to substantial performance losses from temperature glide when the evaporator operates in parallel flow.
- Condenser performance decreases under all test conditions for R-474A and R-479A, with the largest reduction occurring under intermediate conditions.
- Likely causes of condenser performance degradation include: (1) lower outlet refrigerant temperature from temperature glide, reducing heat transfer in the subcooled region, and (2) reduced internal heat transfer coefficient due to heat transfer deterioration.
- The observed improvement in evaporator performance across all conditions is attributed to reduced apparent temperature glide caused by the saturation temperature drop from refrigerant pressure drops. However, this improvement is offset by the performance degradation associated with those same pressure drops.

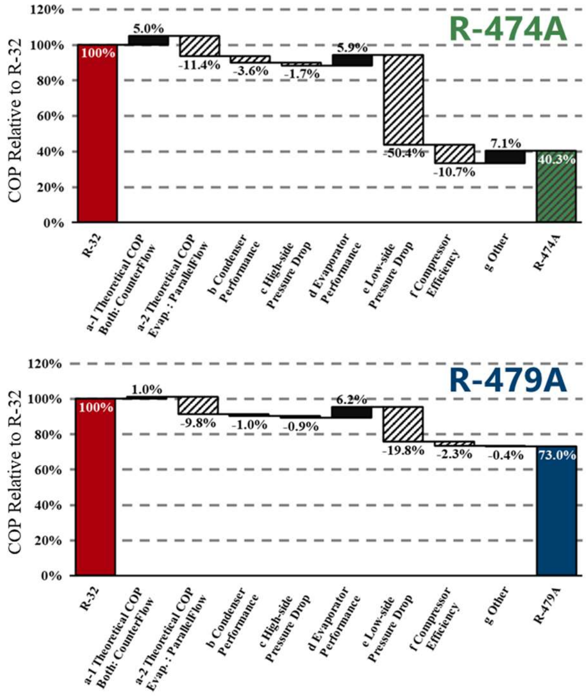

Fig.8 Contribution Analysis of Performance Factors (Cooling Rated)

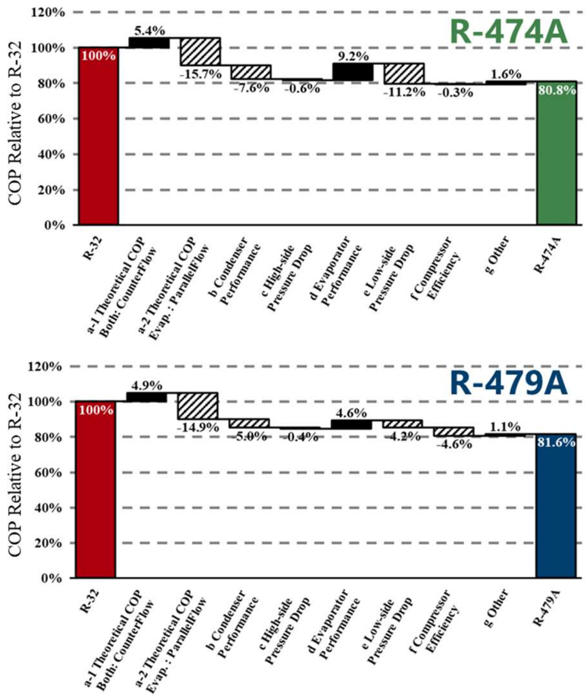

Fig.9 Contribution Analysis of Performance Factors (Cooling Intermediate)

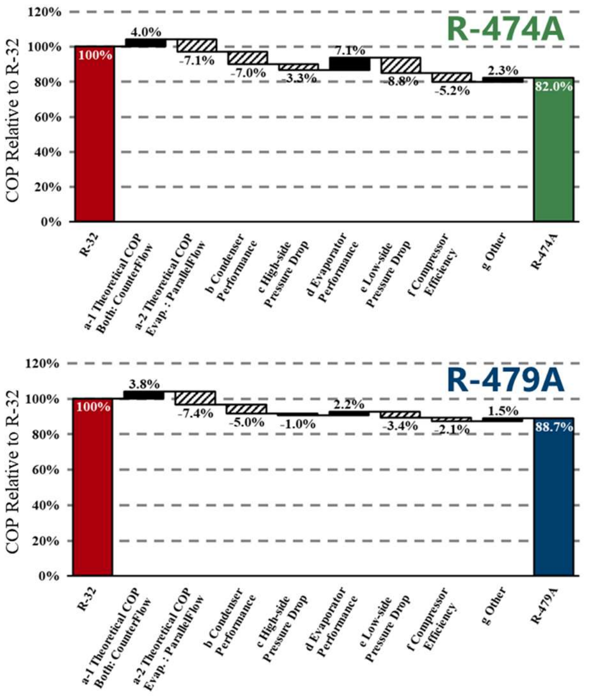

Fig.10 Contribution Analysis of Performance Factors (Heating Rated)

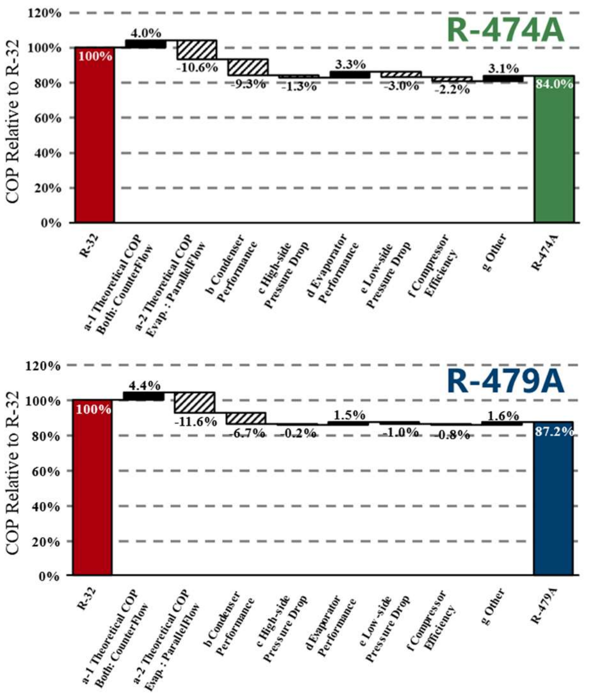

Fig.11 Contribution Analysis of Performance Factors (Heating Intermediate)

PERFORMANCE IMPROVEMENT ESTIMATES FOR LOW-GWP REFRIGERANTS

PERFORMANCE IMPROVEMENT MEASURES

Table 5 summarizes the performance degradation factors identified in the contribution analysis, along with the corresponding measures for their mitigation.

Table 5 Performance Improvement Measures

	Performance Degradation Factor		Performance Improvement Measure
			Adjust Component Parts (Compressor) and Operating Points
I	Increased Refrigerant Pressure Loss	1	Enlarge Gas Line Diameter
		2	Adjust Heat Exchanger Passes
II	Condenser Performance Degradation	1	Enlarge Outdoor Unit (Cooling Rated 5.6 kW)
		2	Enlarge Indoor Unit (1.5x Size)
		3	Enlarge Outdoor Unit (Cooling Rated 9.0 kW)
III	Evaporator Air and Refrigerant Flow Direction (Like Parallel Flow)	1	Ensure Heat Exchanger Counter Flow

RESULTS OF PERFORMANCE IMPROVEMENT ESTIMATION

Performance improvements were estimated by starting from the drop-in conditions of the drop-in performance evaluation tests and incrementally applying the measures listed in Table 5. COP values were estimated for each condition, and APF was then calculated using our in-house refrigeration cycle simulation tool. In addition to R-474A and R-479A, whose drop-in performance was tested on actual equipment, the predicted performance of R-474B and R-479B (Table 2) was also evaluated.

The results of the performance improvement estimation (APF), shown in Fig.12, are summarized below:

- Across all conditions, APF ranked in the order R-479B > R-479A > R-474B > R-474A, with refrigerants having lower GWP showing lower APF.
- For the test unit used in this study, R-474A and R-474B require countermeasures against refrigerant pressure loss to maintain cooling rated capacity.
- With fewer improvement measures applied, refrigerant performance differences were more pronounced.
- Within the measures considered, only R-479B achieved an APF comparable to R-32 without ensuring counterflow heat exchanger configuration. For refrigerants with a large temperature glide, countermeasures addressing the temperature glide are required.

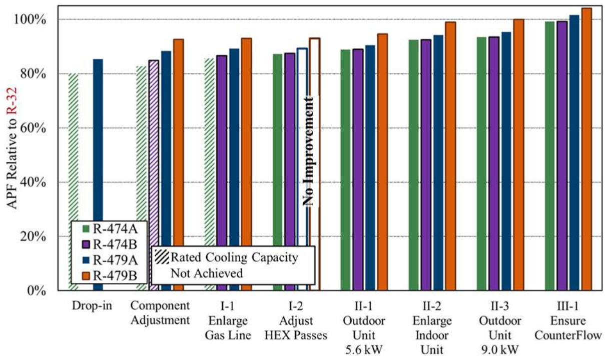

Fig.12 Results of Performance Improvement Estimates

CONCLUSIONS

For the low-GWP refrigerants R-474A and R-479A, drop-in performance evaluation tests for residential air conditioners and performance improvement estimations were conducted. The findings are summarized as follows:

- R-479A showed higher COP and APF than R-474A; however, both refrigerants achieved less than 90% of the performance of the current refrigerant, R-32.
- R-474A exhibited insufficient cooling rated capacity under rated conditions, primarily due to increased refrigerant pressure losses rather than insufficient compressor stroke volume.
- Regarding performance degradation, the major factor under rated conditions was the increase in refrigerant pressure loss, while under intermediate conditions it was the effect of temperature glide.
- Estimated APF values, when cumulative improvement measures were applied for each refrigerant, the APF was highest in the order: R-479B > R-479A > R-474B > R-474A. A tendency was observed in which refrigerants with lower GWP had lower APF. Moreover, the smaller the specification changes from the current system, the greater the performance differences among the refrigerants.
- Only R-479B achieved an APF comparable to R-32 without ensuring the heat exchanger permanently counterflow.

NOMENCLATURE

$COP_{\mathrm{theoretical}}$	:	Theoretical COP
$COP_{\mathrm{R\text{-}32}}$	:	Measured COP of R-32
$COP_{\mathrm{Drop\text{-}in}}$	:	COP in drop-in evaluation
R_{COP}	:	COP ratio
Δh_{ihex}	:	Specific enthalpy change of refrigerant in indoor heat exchanger, $\mathrm{kJ \cdot kg^{-1}}$
Δh_{comp}	:	Specific enthalpy change of refrigerant during compression process, $\mathrm{kJ \cdot kg^{-1}}$

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